# Performance Optimization of a Two-Stage Piezohydraulic Servovalve

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# Abstract

This paper describes the performance optimization of a two stage piezohydraulic servovalve developed for use in aerospace. The valve uses a piezoelectric multilayer actuator in the pilot stage and a conventional main stage spool. The actuator moves a deflector which directs a jet to to create a differential pressure at the pilot control ports which drives the main stage spool. A mechanical feedback wire provides position feedback of the main stage spool to the deflector. The valve has been developed in an attempt to reduce servovalve manufacturing cost.

From a simplified model it can be shown that the maximum spool displacement and the frequency response of the valve are directly influenced by the relative stiffness of the piezoelectric actuator and the feedback wire. In this paper, the model is used to predict this design trade-off and hence optimise the performance of the valve. Two versions of the valve are tested to validate the prediction method.

KEYWORDS: servovalve, piezoelectric actuator, deflector jet, bimorph

# 1. Introduction

Typically a two stage servovalve has a pilot stage employing a torque motor coupled to a hydraulic amplifier with mechanical feedback of the second stage spool position. The hydraulic amplifier is generally a nozzle-flapper, a jet pipe or a deflector jet. The number of parts, tight tolerances and set up costs associated with the torque motor assembly and supporting flexure tube, add to the cost of manufacture. This paper is concerned with the development and optimization of a deflector jet servovalve driven by a piezoelectric bimorph using mechanical feedback. The concept reduces the part count and avoids the need for a flexure tube.

Several ideas for integrating piezoelectric actuators in valves have already been investigated by other researchers. Bang et al. /1/ developed an electrical feedback twostage servovalve using a high power piezoelectric stack actuator to control the flapper in the pilot stage. Due to high operating voltages (250V), compensation for hysteresis and thermal expansion of the piezoelectric elements had to be considered. Milecki /2/ developed a servovalve with a piezoelectric bimorph actuated nozzle-flapper pilot stage. A conventional bimorph actuator produces insufficient force to work against a feedback spring, so electrical feedback was used. Sedziak /3/ reported a similar pilot stage design, where the second stage spool was spring loaded on either end to produce proportional flow – the hysteresis was ≈13%. Karunanidhi et al. /4/ developed a two stage piezohydraulic servo valve with a stroke amplified piezoelectric stack actuator. The design required high operating voltages (150V). Stacks are also not suitable for continuous operation at high frequency due to high current requirements Reichert /5/ developed an electrical feedback valve with four and overheating. piezoelectrically driven poppet valves forming at H-bridge first stage. Hysteresis effects of the actuators were avoided by using a charge amplifier instead of conventional voltage amplifiers.

Recently, integrated multilayer bimorph actuators have been developed. These actuators comprise thin active piezoelectric layers (e.g.  $20\mu$ m) co-fired together with internal electrodes. This reduces the operating voltage (to e.g. 30V) while maintaining high field strength, and increases the converted mechanical energy per volume of piezoelectric material /6/. In addition they have an inactive layer of ceramic encapsulating the actuator which provides humidity resistance. This facilitates submerged operation.

The piezohydraulic servovalve (PHSV) described here uses a multilayer bimorph to drive a deflector jet pilot stage. The new multilayer technology, coupled with the very low deflector jet flow forces, provides sufficient force capacity to retain low cost mechanical feedback. Mechanical feedback is also considered inherently safe, and so is always used for aerospace flight control valves.

#### 2. PHSV operating principle and modelling

Figures 1 and 2 show the construction of the valve. The PHSV operates as follows:

- 1. At the null position (no voltage to the bimorph) the flow from the deflector impinges equally on the control ports, so that the pressure on the main stage spool ends are equal.
- 2. When a voltage is applied to the bimorph, the electric field generates a bending moment along its length. The actuator bends and moves the deflector.
- 3. The displacement of the deflector differentially directs the jet of fluid towards one of the two control ports, thus increasing the pressure in that port. This creates a pressure imbalance across the main spool. This differential pressure moves the spool in the opposite direction to the movement of the deflector.
- 4. As the spool begins to move, it pulls the tip of the feedback wire with it. This generates a restoring force which re-centres the deflector. When the restoring force due to spool movement is equal to the bimorph force at the deflector, the spool stops at that position.

A detailed non-linear dynamic model has been derived for the valve /7,8/. However, a first order linear model is useful for optimizing the valve. This is developed below.

The force on the deflector is:

$$F_d = k_v v - k_{ff} x_d - k_{ds} x_s \tag{1}$$

where *v* is the voltage applied to the bimorph,  $x_d$  is the deflector displacement and  $x_s$  is the spool displacement. The coefficient  $k_{ff}$  is the flow force on the deflector, which is approximately proportional to displacement,  $k_v$  can be found from the blocking force of the bimorph, and  $k_{ds}$  is found from the stiffness of the feedback wire /7/. Thus if  $k_d$  is the stiffness at the deflector:

$$F_d = k_d x_d \tag{2}$$

then

$$x_d = \frac{k_v v - k_{ds} x_s}{k_d + k_{ff}}$$
(3)

neglecting inertial and damping forces on the deflector. Considering the spool, and neglecting compressibility, and leakage:

$$A_{s}\dot{x}_{s} = c_{q}x_{d} \tag{4}$$

where  $A_s$  is the spool end area, and  $c_q$  is the pilot stage flow gain. Combining equations (3) and (4) gives the transfer function:

$$\frac{X_s}{V} = \frac{K_{ss}}{\frac{1}{\omega_b}s + 1}$$
(5)

where

$$K_{ss} = \frac{k_v}{k_{ds}}$$
 and  $\omega_b = \frac{c_q k_{ds}}{A_s (k_d + k_{ff})}$  (6)

Using the Mark-1 valve parameters (see Table 1) at 140bar supply pressure gives:

$$K_{ss} = 6.93 \times 10^{-3} \text{ mm/V}$$
 and  $\omega_b = 284 \text{ rad/s}.$ 

Thus the -3dB bandwidth of the valve predicted by this model is 45Hz, which is similar to the measured value shown in **Figure 3**. The maximum spool displacement predicted by the model at 30V demand is approximately 0.21mm, and this is consistent with the measured steady state gain of -43dB.



Figure 1: Schematic cross section of the PHSV



Figure 2: Cross section of the PHSV prototype

Bimorph	Operating voltage (V)	±30
	Nominal free displacement (µm)	±80
	Dimensions LxWxT (mm)	12x9.6x0.65
	Blocking force(N)	±2
	Electrical capacitance (µF)	6.8
	Bimorph flexural modulus (Nm <sup>2</sup> )	0.0132
Valve	Deflector length (mm)	6
	Feedback wire length (mm)	12.85
	Feedback wire flexural modulus(Nm <sup>2</sup> )	0.0019
	Spool end area (mm <sup>2</sup> )	34.3

Table 1: Mark-1 PHSV parameters



Figure 3: Frequency response of the Mark-1 PHSV at 140bar supply pressure and maximum amplitude

#### 3. Design optimization

As shown in /7/, the steady state gain (and hence maximum spool displacement) and bandwidth frequency are functions of valve parameters as follows:

$$K_{\rm ss}(k_r, L_r, L_{\rm dr}, L_{\rm f}) \tag{7}$$

$$\omega_b(k_r, L_r, L_{dr}, k_q, A_s, E_f I_f, L_f)$$
(8)

where  $k_r$  is the ratio of bimorph flexural stiffness over feedback wire flexural stiffness  $E_{t}I_{f}$ ,  $L_r$  is the ratio of bimorph length over feedback wire length  $L_f$ , and  $L_{dr}$  is the ratio of rigid deflector length over feedback wire length. The steady state gain is also dependent of the piezoelectric strain constant and the layer thickness.

Assuming the deflector jet, feedback wire and main stage spool parameters are fixed, and for a specified supply pressure, the gain and bandwidth are dependent on:

- the length ratio, L<sub>r</sub>,
- the flexural stiffness ratio,  $k_r$ ,
- the rigid deflector length ratio, *L*<sub>dr</sub>.

Figure 4 shows the effect of varying the length and flexural stiffness ratios of the prototype valve, while keeping the deflector guide length the same. The model for

140bar supply pressure is used and the steady state gain is expressed as the maximum spool displacement, i.e. displacement with maximum applied voltage. The effect of changing  $L_{dr}$  is discussed in /7/.



**Figure 4:** Design trade-off for the bandwidth and steady state gain at 140bar supply pressure and maximum applied voltage

In order to help validate the variation in performance predicted in Figure 4, a second valve was constructed. The Mark-2 prototype has a greater bimorph free length, increased from 12mm to 20mm. As shown on Figure 4, the simple model now predicts a bandwidth of 59Hz, and a spool displacement of  $\pm 0.16$ mm. This is consistent with the measured result shown in **Figure 5**.



Figure 5: Frequency response of the Mark-2 PHSV at 140bar supply pressure and maximum amplitude

#### 4. Discussion

A physical interpretation of Figure 4 follows. For a given  $k_r$ , the bandwidth increases with  $L_r$ . This is because increasing  $L_r$  reduces the relative stiffness of the bimorph to the feedback wire. The spool travel (steady state gain) is reduced due to the relatively increased feedback wire stiffness. This reduces the time taken by the feedback wire to centralise the deflector. Hence the bandwidth is increased.

For a given  $L_r$ , the steady state gain increases with increasing  $k_r$ . This is because increasing  $k_r$  increases the relative stiffness of the bimorph to the feedback wire. The spool travel is increased due to the relatively low stiffness of the feedback wire. This increases the time required by the feedback wire to centralise the deflector. The bandwidth of the valve reduces.

In this case, as  $L_r$  value increase above about 2 the maximum force generated by the bimorph reduces below approximately 1N, and the influence of flow force on the bandwidth will become significant. A greater flow force will tend to reduce the deflector displacement and thus reduce the pilot stage flow and the bandwidth. Hence there is a limit to the increase in bandwidth with increase in bimorph length. In addition a long bimorph may introduce bimorph-feedback wire resonance issues.

In this case, doubling  $L_{dr}$  increases the bandwidth of the valve by approximately 10Hz /7/. However, the steady state gain is reduced. The deflector guide length acts as an amplifier to the bimorph tip deflection. The increased deflector displacement increases the spool velocity and thus the bandwidth of the valve spool. However, the restoring force required by the feedback wire to centralise the deflector is reduced. Hence the spool travel is reduced. This reduces the steady state gain of the valve.

# 5. Conclusions

In this research a novel pilot stage actuator fitted to a conventional deflector jet servovalve was investigated. The torque motor assembly was replaced by a multilayer bimorph actuator. A mechanical feedback wire was used for proportional flow control. The bimorph was directly coupled to the feedback wire for submerged operation. This considerably simplified the pilot stage manufacture and reduced the part count which could lead to cost savings.

A simple (first order) dynamic model is presented. The main factor that gives a high bandwidth is a high ratio of deflector to spool movement, which is dependent on the relative stiffnesses of bimorph and feedback wire. Conversely, if a large spool displacement is required to centralize the deflector following a step in applied voltage, the steady state gain is large. A design optimization plot is presented showing that these conflicting requirements must be taken into account when sizing the valve.

Two prototype valves were developed and tested to validate the design trade-off predictions. The -3dB bandwidth of the prototype valve at 140bar supply pressure and maximum applied voltage amplitude were approximately 40Hz and 60Hz respectively. This is in reasonable agreement with the predictions.

# 6. Acknowledgements

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# 8. Nomenclature

As	spool end area	m²
E <sub>f</sub> I <sub>f</sub>	feedback wire flexural stiffness (product of Young's modulus and 2 <sup>nd</sup> moment of area)	Nm <sup>2</sup>
F <sub>d</sub>	total force on deflector	Ν
<i>k</i> <sub>d</sub>	stiffness at deflector	N/m
k <sub>ds</sub>	force on deflector due to spool movement	N/m
<i>k</i> <sub>ff</sub>	additional deflector stiffness due to flow forces	N/m
$c_q$	deflector flow gain	m²/s
K <sub>ss</sub>	steady state gain	mm/V
<i>k</i> <sub>r</sub>	flexural stiffness ratio (bimorph over feedback wire)	
L <sub>dr</sub>	ratio of deflector to feedback wire length	
L <sub>f</sub>	feedback wire length	m
L <sub>r</sub>	ratio of bimorph to feedback wire length	
V	applied bimorph voltage	V
Xd	deflector displacement	m
Xs	spool displacement	m
ω <sub>b</sub>	bandwidth	rad/s